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Performance of Automatic Compressor Valves

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PERFORMANCE OF AUTOMATIC COMPRESSOR VALVES

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INTRODUCTION

A previous paper (1) has reported measurements of dynamic force on a disc valve, during the opening of the valve, and has shown that significant differences are to be found between this true dynamic force and the corresponding force measured under steady state conditions at increments of valve displacement. In this paper, the authors describe an investigation of the velocity field around the same disc valve, again measured dynamically during the rise of the valve. The measurement uses miniature 45° sensor probes, in a constant temperature hot wire anemometry technique, to define the instantaneous velocity vectors, and analysis of the signals from the wires is by a fully calibrated method developed by the authors for this project. It is shown that entrained gas from above the valve plays a significant part in the development of flow patterns, and that the development of eddies from discontinuities in the surface of the valve seat (eg from ring seats) is, on a time scale, of the same order as that of the rise time of the valve. Analysis is confined to the assumption that flow is in radial planes.

Test Rig and Instrumentation - General

When measuring valve force (1) it was found to be necessary to arrange a moving plenum chamber and seat, with a fixed valve, because of the sensitivity of force transducers to acceleration. When measuring velocity, on the other hand, it is obviously impractical to arrange the velocity sensor in a position occupied by the seat before initiation of the opening phase, and the rig was therefore arranged as shown in Fig. 1, with a fixed plenum chamber and seat, and a moving valve. The energy for a fast rise time was stored in a powerful spring and the valve held shut against this spring by means of an electro-mechanical actuator connected to a DC voltage. This arrangement gave a rise time for the valve of approximately 5 ms, which was felt to afford a reasonable comparison with some compressor valves.

Valve displacement was continuously measured by a capacitive transducer. The fully open reading was checked by slip gauge.

Initial and final pressures in the plenum cylinder were noted from a calibrated pressure gauge, while pressure variation during the tests was recorded from a piezoelectric pressure transducer at a site corresponding to the pressure gauge tapping. It was found that cylinder pressure did not vary significantly during the rise time of the valve.

Signals for the calculation of velocity vectors were taken from a sub-miniature 45° sensor probe coupled to a hot wire anemometer bridge. The sensor position was adjusted radially and vertically by means of a positioning device using two micrometer heads. Data for position were obtained by means of a measuring microscope, relative to the valve periphery for radius, and relative to the face of the seat for height, taking in each case the centre of the 45° sensor. Velocity being the main measurement of the tests, is discussed at greater length in a separate section.

The test outputs of the various transducers were recorded on a digital storage oscilloscope, then transferred to an instrumentation tape recorder for subsequent analysis. During analysis, the signals were re-transmitted from the tape recorder into the digital storage oscilloscope in a piece wise fashion, then passed through a digitiser and interface to a mini-computer for storage in a data matrix. Subsequently, computer analysis was used to extract the magnitude and direction of the velocity vector, with relevant valve displacement and time for a grid of sixteen positions. The grid is shown in Fig. 1. Data are available at time intervals of the order of 25 microseconds. A few examples of these diagrams are shown in Fig. 3.

Interpretation of Cross Wire Anemometer Signals

Dual sensor cross-wire probes as supplied by manufacturers tend to be bulky relative to the dimensions of the valve used in these tests. To secure maximum definition of position it was therefore decided to use a single 45° sensor sub-miniature probe for all readings, obtaining an output equivalent to a cross wire reading by rotating the probe through 180° about the axis of its shank for readings which otherwise corresponded in sensor position, cylinder pressure, etc. This relies on repeatability of the conditions of the test, but has the advantage of eliminating differences in the characteristics of the sensor and its anemometer bridge, which are sources of error in using dual filament probes. The parameters which must be repeated are initial cylinder pressure and the displacement/time curve. Of these, the former can be set with some accuracy, while analysis showed that in all tests the displacement/time curve repeated to an accuracy within the low noise level of the electronic instrumentation. It is therefore felt that results from the single sensor method used are as good as, or possibly better than, those which would obtain for a dual sensor probe.

The probe was calibrated in the parallel jet of a specially constructed nozzle calibrator, at a series of speeds and at angles from -100° through zero to $+100^{\circ}$.

The findings of this calibration were:

(a) that even in the well defined plus to minus 45° zone, mathematical models thus far proposed are not adequate to represent signal behaviour, while at angles beyond this zone, in which prong interference can be a factor in flow, only calibration of the type used can give a true representation of the flow,

(b) the characteristic shape of the signal versus angle curve is essentially the same for each calibration gas speed and can thus be represented for all speeds within the range by the same suitably chosen polynomial.

Using this calibration, a curve of angle versus the ratio V1/V2 was fitted as a third order polynomial, where V1 and V2 are voltage outputs from the anemometer bridge during tests, for corresponding sensor orientations at 90° to each other. This was found to be single valued over the range. Hence the angle between the gas flow and the probe axis is uniquely determined. Having found the angle of flow, it is merely a matter of computation to calculate the total velocity from the calibration curves, using their common shape to simplify this analysis.

For each test, there is thus produced, the magnitude and direction of the velocity vector corresponding to the probe position, valve displacement, time from initiation and cylinder pressure. Some further computation gives, for each grid position and pressure, graphs of velocity, angle of flow and displacements, to a base of time. A sample of this output is shown in Fig. 2. To illustrate the changing behaviour of the flow as time goes on, velocity vectors at all grid points for corresponding times were subsequently extracted and these are shown in Figs. 3(a) to 3(f).

Test Rig Data

Valve diameter = 8.40 mm Port diameter = 6.35 mm
Measurement grid positions (16 positions in all)
X (from valve periphery), mm 0.48 2.08 5.26 11.61
Y (from face of valve seat), mm 1.17 2.08 3.00 3.96
Initial cylinder pressure for all reported data $= 41.5\ \mathrm{kN/m^2}$
Hot wire sensor length = 0.45 mm at 45° to shank

DISCUSSION

Much of the work for this paper has been devoted to the development of satisfactory methods of data acquisition and analysis, and it is clear that further work is required before it can be claimed that dynamic flow during the opening phase of this simple disc is fully understood.

The physical size of the smallest hot wire probe obtainable is such that the minimum height position in the grid loses much valuable data close to the seat. It may be that investigation with a larger valve is required. Nevertheless, some interesting trends appear in the diagrams obtained.

Fig. 2 shows one sample of the diagrams for angle, velocity and displacement at one grid position and initial pressure. This particular diagram is for the maximum radius, at the second height from the seat, and shows clearly the delay of approximately 2.5 milliseconds between the inception of valve displacement and the arrival of the front of the released gas at the measuring station. Following arrival of the front, the angle of flow is -60° for about half a millisecond, then moves rapidly to the region $+20^{\circ}$ to $+70^{\circ}$. The convention for these diagrams is that all angles are relative to horizontal, minus indicating upwards, plus downwards. After inception of flow, velocity climbs rapidly to a maximum value of about 18 m/s, before dropping back to about 5 m/s at the fully open displacement. It should be noted here that the fully open displacement does not by any means mark the end of transient conditions and that further analysis is necessary between this time and final steady state conditions. The work carried out for this paper indicates that in some cases this may occupy a total time of the order of 25 to 30 milliseconds.

At this stage, it may be well to consider the nature of the phenomenon under investigation. The discharging gas is subject to start from rest, and additionally to the effects of a continuously varying valve displacement. The former of these conditions implies that initial flow is in a laminar mode, requiring finite time for the development of eddies both at surfaces of velocity discontinuity, with consequent high rates of shear, and at boundary discontinuities such as the step in the seat (representing a ring seat). The latter condition of varying valve displacement implies not only an increasing rate of discharge at source, but also a variation in the angle of the conical surface of velocity discontinuity. Because of distance velocity lag, it follows that at any instant during the opening phase, the line of velocity discontinuity in a radial plane is not straight, but curves downwards with increasing radius.

Figs. 3(a) to 3(f) are vector diagrams prepared by computer from the matrix store of all such information as is exemplified by Fig. 2. Lacking information on the extent of the curvature of the line of discontinuity discussed above, a notional straight line of velocity discontinuity has been added to each diagram as an extrapolation of the line joining port edge to valve edge, to give some idea of the separation of flow zones.

At 2.2 ms (Fig. 3(a)), with displacement 0.13 mm, there is some indication that the valve edge is behaving in part as a circular line source, with spread of fluid as a broad jet occupying most of the available radial plane. This is not, however, completely so, for there is already evidence of entrainment, showing small vectors down the side of the valve, turning towards the jet direction as the edge is approached. It is also clear that the escaping gases have a well defined front.

At 2.7 ms and 0.19 mm displacement (Fig. 3(b)), there is evidence of the establishment of a whole field of entrainment above the velocity discontinuity, and this trend is continued throughout the remaining diagrams. It is noticeable, however, that entrainment vectors pass through a maximum amplitude before reverting to relatively small values in the final steady state (100 ms) diagram (Fig. 3(f)). It seems likely that the strong entrainment is the result of high rates of shear consistent with a predominantly laminar regime and that the final small vectors indicate a fully developed turbulent zone between the two flow domains. At this time (2.7 ms) vectors on or below the discontinuity line are consistent with the variation in velocity to be expected from the differing measurement radius, again suggesting laminar flow with only small losses.

At 3.1 ms (Fig. 3(c)), the velocity at position A has become small compared to the velocities achieved at greater radii. This is accompanied by a reduction in the entrainment vector, possibly indicating that eddies are being established on the line of discontinuity at this radius.

The diagrams for 3.7 ms and 4.2 ms (Figs. 3(d) and 3(e)), show flows at the largest radius which are clearly consistent with the growth of a large eddy at the seat step. Since this is not shown at 3.1 ms, it may be argued that the time required for the establishment of this eddy is around $3\frac{1}{2}$ milliseconds.

The final diagram at 100 ms (Fig. 3(f)), is prepared from time averaged velocity and angle information over a suitable period within the range of time at which transient effects had disappeared. It is interesting to note that in steady flow, the large eddy beyond the step in the seat has been largely suppressed and that flow is essentially parallel to the seat at larger radii.

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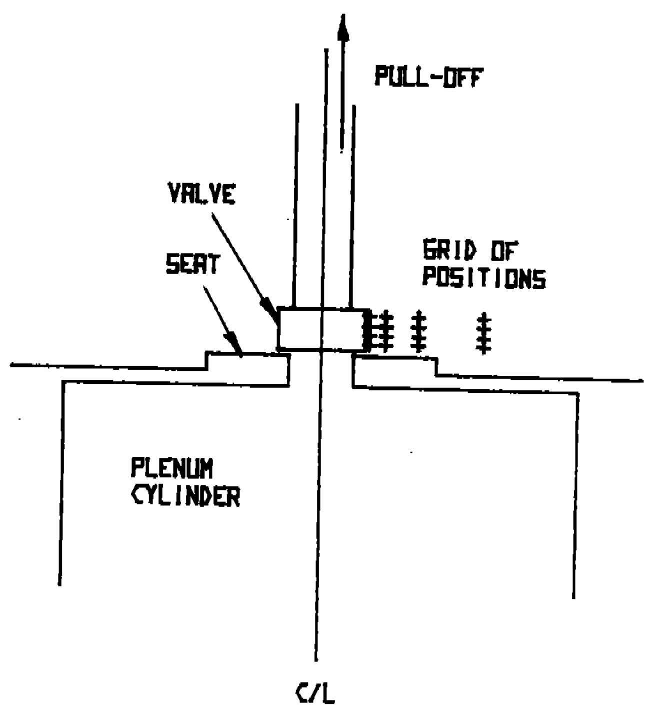

FIG 1 ARRANGEMENT OF TEST RIG

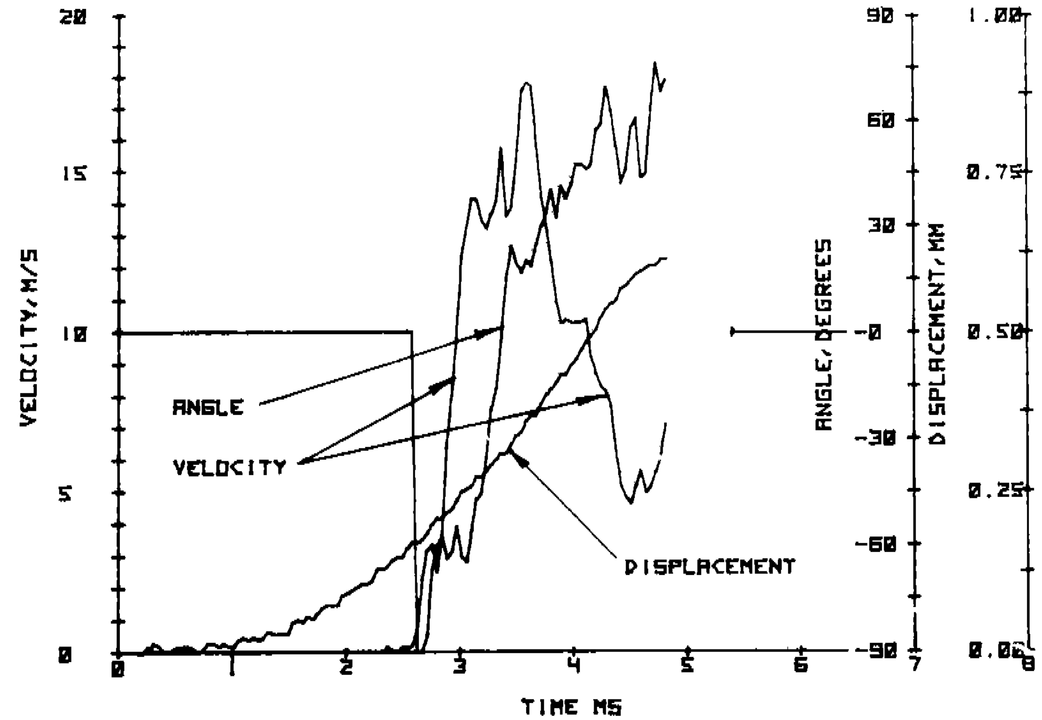

FIG.2 ESCAPE VELOCITY & ANGLE
POSITION, X= 11.61MM, Y= 2.00MM P.D.= 41.5KN/SQM

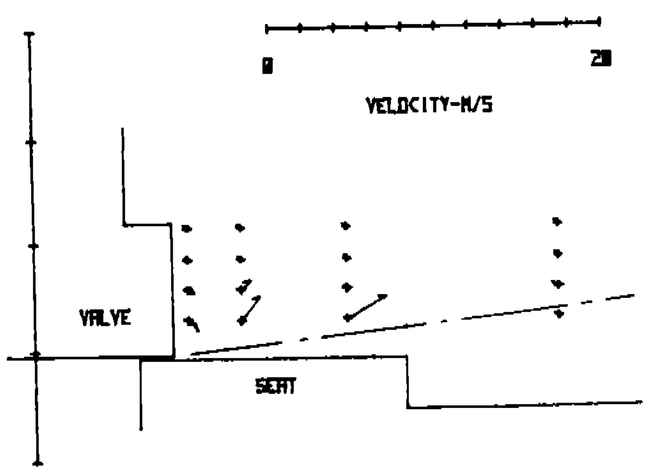

FIG 3(A) DISP 0.13 MM TIME 2.2 MS
VELOCITY VECTORS

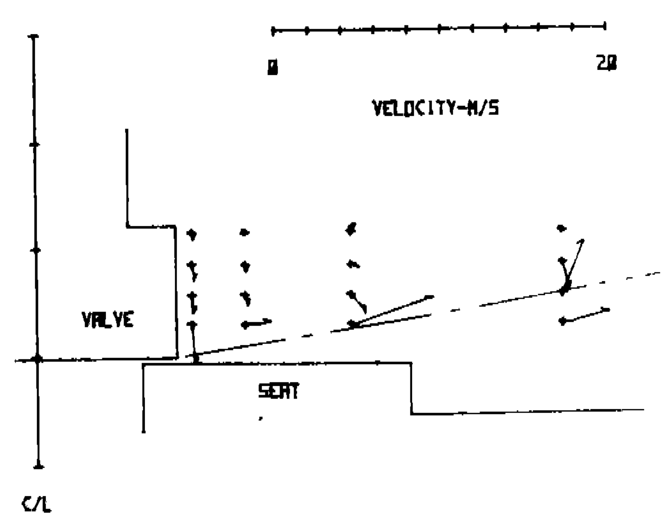

FIG 3(B). DISP 0.19 MM TIME 2.7 MS
VELOCITY VECTORS

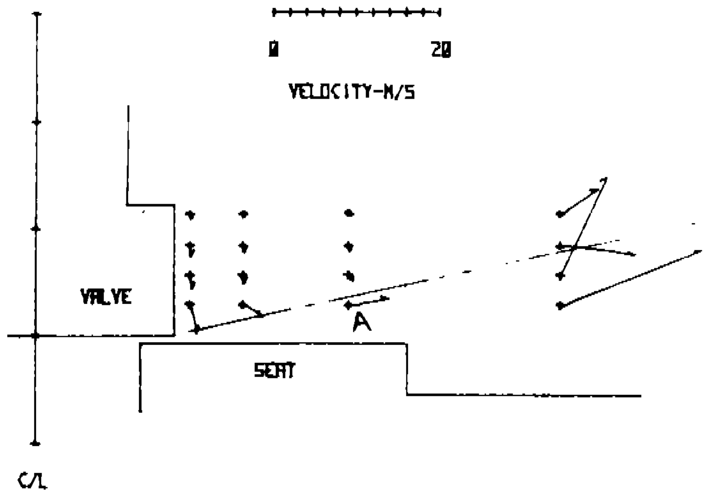

FIG 3(C) DISP 0.25 MM TIME 3.1 MS
VELOCITY VECTORS

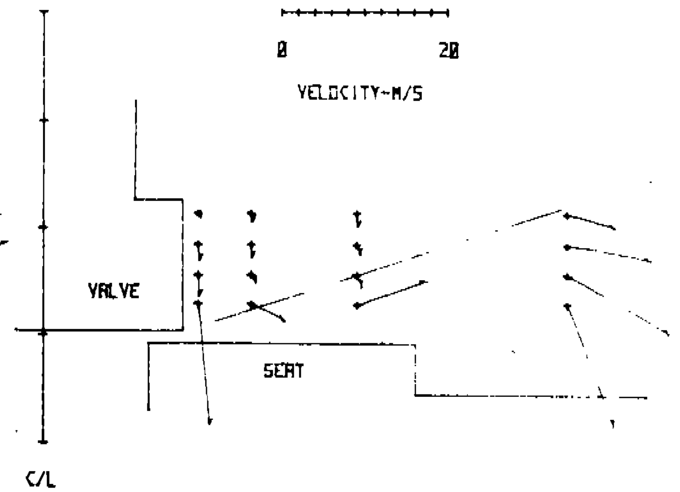

FIG 3(D). DISP 0.30 MM TIME 3.7 MS
VELOCITY VECTORS

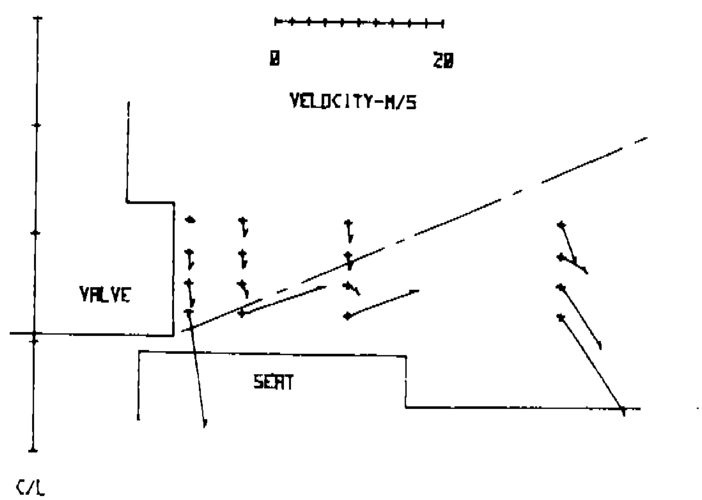

FIG 3(E) DISP 0.50 MM TIME 4.2 MS
VELOCITY VECTORS

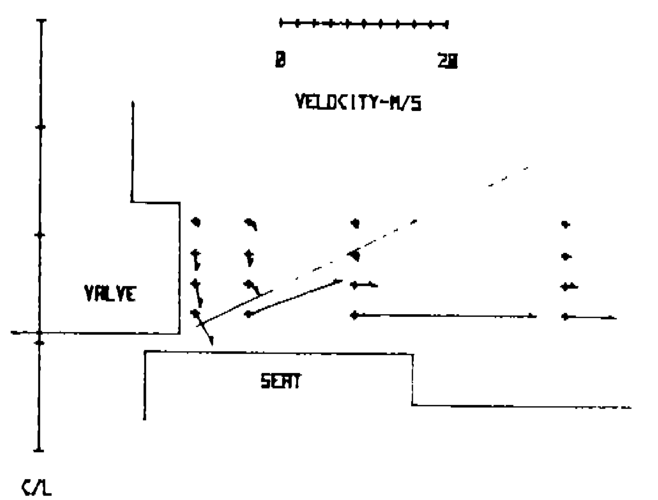

FIG 3(F) DISP 0.50 MM TIME 100.0 MS
VELOCITY VECTORS